VIABILITY OF LARGE HIGH-SPEED DISPLACEMENT HULLS

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An appraisal is made of the technical feasibility of developing a 50-kt displacement-hull type ship capable of delivering a 12,000 LT payload over a distance of 9000 nm. Initial parametric analysis concludes that this ship will have a length of 1500 ft, a displacement of 64,000 LT, a draft of 30 ft, a length/beam ratio of 12.6 and a transport efficiency factor nearly twice that of any existing 50-kt ship. Further, it will require nearly 600,000 hp, which is substantially larger than that in any existing marine vessel.

Using basic hydrodynamic principles and published model test data, it is shown that the performance, seakeeping, the transverse stability and the lateral dynamic stability will be satisfactory.

A design synthesis program called PASS (Parametric Assessment of Ship Systems) is used to develop and evaluate a notional design for this concept. It is concluded that the concept can support the propulsion plant, fuel and payload, while still satisfying basic structural and stability requirements.

The structural integrity of the hull is further evaluated using the SafeHull structural program developed by the American Bureau of Shipping (ABS). The structural integrity and weight fraction of the hull also appears to be satisfactory and reasonable.

Operational and economic aspects of this long ship are not considered at this time. This evaluation has shown that, at this point, it appears to be a feasible concept. However, a number of research and development issues are identified and must be evaluated if the concept is to be considered further.

INTRODUCTION

Development of marine vessels capable of transporting large cargo tonnage at speeds in excess of 50 kts continues to be a challenge to the naval architect. While 50-kt craft do exist they are of modest size, require very large horsepower per ton of displacement, and cannot transport large payloads over intercontinental routes. For the most part the hullforms of these existing high-speed marine vehicles include SES, catamarans, planing monohulls, hydrofoils, and semi-displacement types. They operate at speed-length ratios $\left(V_k / \sqrt{L}\right)$ substantially greater that 1.3 where the wave-making resistance is large and where the dynamic lift replaces buoyant lift at a cost of added induced resistance.

The most current project in this arena is the FastShip Atlantic program. This is an 860-ft, 40-kt ship with a cargo capacity of 10,000 LT. It is a semi-displacement type hull operating at a speed-length ratio of about 1.4.

Basic principles of naval architecture show that a displacement type hullform is the most efficient (smallest horsepower/ton of displacement) type of marine vessel, providing that it operates at a speed-length ratio less than 1.3. In this speed range wave-making resistance is small and the ship weight is supported mainly by "free-of-charge" buoyancy. Because of their large size (and, hence, large cargo capacity) displacement type ships are the preferred hullform for maritime transport. The present economical maximum speed of these vessels is approximately 30 kts. For typical ship lengths of 900 ft, their speed-length ratio is 1.0, well

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below the accepted upper limit of 1.3 for displacement hulls.

In 1961, MARAD conducted a series of model tests on high-speed displacement hulls. These tests were followed by the Series 64 tests conducted by the U.S. Navy in 1965. Both of these programs proved the validity of the hydrodynamic theories for high-speed displacement hulls.

The objective of the present study is to examine the technical practicality of developing these efficient hullforms to operate at speeds of 50 kts. For a maximum speed-length ratio of 1.3, the 50-kt displacement ship will have a waterline length of 1480 ft. Although this is an impressive size, it would not be the longest ship ever built. In 1979, the Sumitoma Shipyard in Japan, actually constructed a 1500-ft long tanker (named the Happy Giant) for a Norwegian company, but its speed was only 13 kts. This current study can also be viewed as the next step beyond FastShip Atlantic. It is emphasized that the present study is a feasibility analysis to determine if such a concept is viable from an engineering standpoint and, as a by-product, to possibly define the upper limits of speed/size combination for displacement type hulls. Potential commercial or military applications are not considered at this time. As expected, the power requirements of this ship will exceed those in any existing marine vehicle. This, as well as anticipated severe structural requirements, presents a major challenge to the naval architect.

This paper defines the basic geometric and hydrodynamic features of a displacement type hull; its relation to the geometry of other hull types; and suggested limits of application of each type as a function of speed-length ratio. Model test results are used to define the resistance and seakeeping of the 50-kt displacement hull. Further, the stability and maneuvering characteristics are discussed.

Naval architecture features such as power plant installation; propulsors; structural design; weight fractions; cargo weight and space limits; range; fuel consumption; ship systems; etc., are analyzed using the algorithms contained in a design synthesis software program called PASS (Parametric Assessment of Ship Systems). A point design is developed and potential problem areas and research and development needs are identified for future development of this concept.

Additionally, the basic structural design developed by PASS is evaluated by the American Bureau of Shipping using their SafeHull program. Alternate structural arrangements are also described.

HULLFORM DEPENDENCE ON SPEED-LENGTH RATIO

The motion of a ship on the water surface generates gravity waves that travel at the ship speed and have a wave length that is proportional to the square of the ship speed. These surface waves have a fixed relation between their speed and their wavelength. In English units the wave speed in knots divided by the square root of the wave length in feet is always equal to 1.34 (except in shallow water). The speed-length ratio of a displacement vessel is similarly defined as its speed in knots divided by the square root of its waterline length in feet. Therefore, when a vessel travels at a speed-length ratio of 1.34 it generates waves whose wave length is equal to the waterline length of the ship. This identifies the upper limit of operation for "true displacement ships". The reasons for this are as follows:

For speed-length ratios less than 1.0 the displacement vessel spans two or more of its self-generated wave crests (see Figure 1). The change in hull draft and trim relative to the static condition are minimal and the power requirements are modest since the hydrodynamic resistance is mainly due to frictional forces on the hull. The geometry of the hullform is characterized by round bilges and a tapered stern. The stern is the aft terminus of the buttock lines that are curved upward toward the water surface to prevent flow separation at the transom. As the ship speed increases, the wavelength increases until at a speed-length ratio equal to 1.34 the wave length is equal to the ship waterline length. Throughout this speed range the generated surface waves are of relatively small amplitude so that the resistance due to wave generation is small. At still higher speed-length ratios, however, the generated wave is larger than the ship length causing the hull to trim up and to literally climb up the back of its self-generated wave (see Figure 1). In addition, the increased local velocities caused by the rounded hullform sections of the displacement hull result in negative bottom pressures that result in an increase in draft and trim by the stern. All these effects accumulate and result in a rapid increase in resistance as a displacement hull is driven at speed-length ratios greater than 1.34. This is sometimes referred to as the "wall of resistance" for a surface ship. Figure 2 presents model data that plots the total resistance coefficient, C_t, versus speed-length ratio. It clearly demonstrates this "wall of resistance" at a speed-length ratio equal to 1.34. This is one of the factors that motivated the present study of a potential 50-kt, 1500-ft displacement ship.

For speed-length ratios greater than 1.34 the hullform should no longer have a “canoe” type stern and is characterized by straight, aft buttock lines terminating in a somewhat submerged and square transom. Round bilge sections can continue to be used in what is referred to as a "semi-displacement" hull. While this form will have a somewhat smaller rate of rise of total resistance with increasing speed-length ratio compared to a displacement hull, it will have substantially higher resistance at lower speed-length ratios.

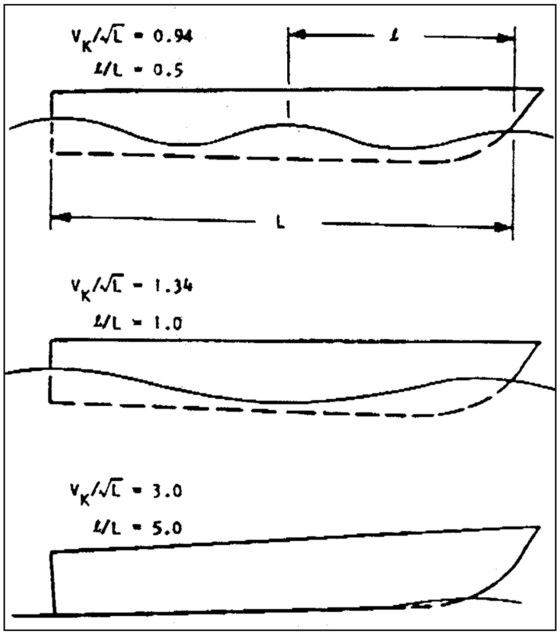

Figure 1. Ship Wave Patterns Versus Speed-Length Ratio

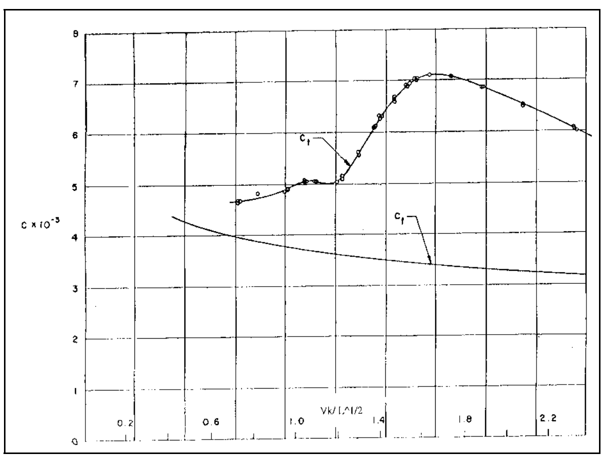

Figure 2. Typical Friction (C_f) and Total Resistance (C_t) Coefficients Versus Speed-Length Ratio

The semi-displacement hullform is the preferred hullform in the speed-length range between 1.34 and 3.0.

For speed-length ratios larger than 3.0, dynamic lift forces are substantial, the hull trims up by the bow and rises above the static floatation level. The hull is now in the planing regime and its geometry avoids convex curvature in both the transverse and longitudinal (except for the bow area) sections in order to prevent the development of negative dynamic bottom pressures. The buttock lines are straight; the transom is wide and submerged and the chines and transom are sharp (no round bilges) to promote complete flow separation from the stern and sides of the hull.

TRANSPORT FACTORS (EFFICIENCY) OF HIGH -SPEED MARINE VEHICLES

The previous discussions have referred to the expected high hydrodynamic efficiency of the displacement type hull. To quantify this observation it is useful to relate the so-called Transport Factor (TF) for the proposed 50-kt, 1500-ft ship to those for existing high-speed marine vessels. A number of variations of the transport factor have been used in past studies. The definition used here is chosen to be consistent with recently published data presented in [1]. The Transport Factor is defined as:

$$TF = \frac{\text{Displacement x Speed}}{\text{Installed Horsepower}} = \Delta\ V/SHP \times 550 = \Delta \times V \times OPC/R_t \times V$$

where:

R_t	=	total resistance, lb
OPC	=	overall propulsive coefficient
SHP	=	shaft horsepower
Δ	=	ship displacement, lb
V	=	ship speed, ft/sec.

Thus, $TF = (\Delta/R_t) \times OPC$

The term Δ/R_t is the "lift-drag" ratio of the ship and is associated with hydrodynamic efficiency. The larger the value, the higher is the efficiency. It is mainly dependent upon the value of the slenderness ratio (LBP/ $Vol^{1/3}$) and the speed-length ratio where LBP is the length between perpendiculars and *Vol* is the displaced volume. The lift-drag ratio increases with increasing slenderness ratio and attains a maximum value at a slenderness ratio approximately equal to 10. In contrast, the lift-drag ratio continues to increase rapidly with decreasing speed-length ratio.

A recent study of high-speed sealift technology [1] provides TF data for a wide range of high-speed vehicle types including the monohull, catamaran, SES, ACV, hydrofoil, airship, and ekranoplane which are currently or were formerly in production. These results are summarized in Figure 3 (taken from [1]) for speeds between 35 and 90 kts. It is seen that the envelope of maximum values of TF decreases rapidly with increasing speed. The expected TF value for the 50-kt, 1500-ft displacement ship, which is the subject of the present paper, is also plotted on Figure 3. It is seen, that at 50 kts, its TF value is nearly double that of the most efficient non-displacement type vessel. It is this observation that primarily motivated the present study of the technical viability of this hullform. Certainly, there remain many questions regarding the commercial viability and operational aspects of this very long ship, the answers to these questions can be the subject of future studies. The subsequent sections of this paper are devoted entirely to technical and engineering considerations and culminate in what is considered to be a notional point design concept.

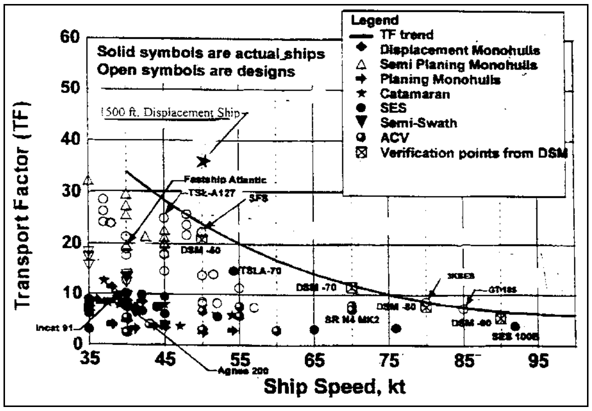

Figure 3. Vehicle Transport Factor (TF) Versus Ship Speed

HYDRODYNAMIC CHARACTERISTICS

Selection of Hullform

In 1961, under sponsorship of the U.S. Maritime Administration, the Davidson Laboratory, Stevens Institute of Technology, conducted a series of model tests to define the smooth water resistance and seakeeping characteristics of a series of displacement hulls. These displacement hulls were tested over a range of speed-length ratios between 1.0 and 2.5 and over a range of displacement-length ratios ($\Delta/(0.01\ LWL)^3$) between 30 and 150 where the displacement Δ is expressed in long tons [2]. In 1965, the David Taylor Model Basin of the U.S. Navy developed and conducted smooth water resistance tests on the Series 64 high-speed displacement hulls [3]. For the Series 64 tests, the speed-length ratios varied between 0.20 and 5.0 and the displacement-length ratio varied between 15 and 55.

The geometric particulars of each hull series are well described in [2] and [3]. Of particular importance is that, for both series, the majority of the hulls have high length-beam ratios, particularly for the lower values of displacement-length ratio. The length-beam ratio for the MARAD series varied from 7.0 to 12 while for Series 64 it varied between 8 and 18. These are extremely fine and narrow hulls. The block coefficient $Cb = (\Delta/LWL \times B \times H \times w)$, where w is the weight density of water, was 0.586 for all the MARAD hulls. The Series 64 hulls had block coefficients of 0.35, 0.45, and 0.55. For both hullform series, the resistance data are presented as ratios of residual resistance/displacement (Rr/Δ), where Rr is in lb) and the wetted surface area is defined. Thus, extrapolation of these model data to prototype ship sizes is accomplished easily. Both data sets are in substantial agreement where test conditions overlap.

In the present study the combination of 50 kts and a limiting speed-length ratio of 1.3 has established the waterline length of the hull to be 1480 ft. A block coefficient of approximately 0.45 was initially assumed since fine waterlines would be consistent with high speeds. Further, a displacement-length ratio of 20 was selected since the residual resistance/displacement ratio is smallest at this loading. Model No. 4804 of Series 64 was selected for this initial study since it most closely matched the desired hull particulars. Figure 4 provides the body plans for this hull. The following is a summary of its principal characteristics:

C_b:	0.45
LWL/B:	12.7
B/H:	4.0
$\Delta/(0.01\ LWL)^3$:	20
$S/(\Delta \times LWL)^{1/2}$:	16.6
$1/2\ \varepsilon$:	5.2 degs

Where:

$1/2\varepsilon$	=	Waterline entrance half angle.
H	=	Draft at full load, ft.
S	=	Wetted surface area of hull, ft^2.
B	=	Waterline beam, ft.
LWL	=	Waterline length, ft.
C_b	=	Block coefficient.

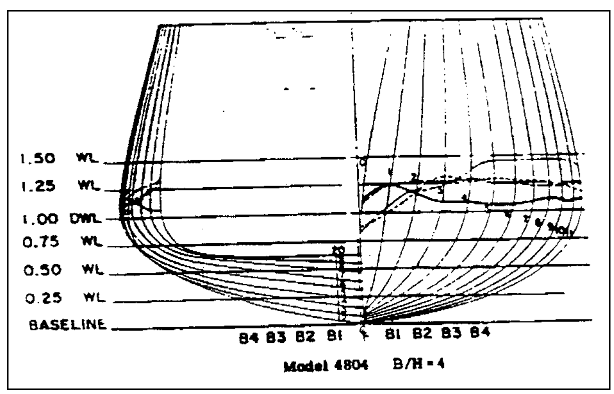

Figure 4. Body Plans for 1500-ft Displacement Ship Concept (Series 64 - Model #4804)

Using these parameters as a starting point, a notional design was developed with the following major geometric features:

LWL	=	1480 ft
B	=	117 ft

H	=	30 ft
Δ	=	63,800 LT
S	=	161,300 ft^2
1/2 ε	=	5.2 degs
C_b	=	0.43.

The development of these characteristics is described in a later section.

Estimated Hydrodynamic Resistance and Powering of Hull at 50 kts

Using the data presented in [3], the initial estimate for residual resistance is:

Rr/Δ	=	15 lb/ton, so that
Rr	=	15 x 63,800 = 957,000 lb

The friction resistance estimate is:

V_k = 50 kts
R_f = 1/2 ρ x (V_k x 1.69)^2 x S x C_f
R_N = V_k x 1.69 x LWL/kinematic viscosity = 9.77 x 10^9
C_f = f (R_N) using ITTC friction line = 0.001175
S = 161,300 ft^2

Thus, R_f = 1,353,000 lb.

Therefore, Total Resistance = R_t = 957,000 + 1,353,000 = 2,310,000 lb.

EHP	=	R_t x V_k x 1.69/550 = 354,900 hp
OPC	=	0.65 (estimated)
SHP	=	EHP/OPC = 546,000 hp.

Allowing for a 10% margin, the shp = 600,600 hp.

As previously mentioned, this power level is extremely high and presents new challenges for the naval architect. Although some development work will be required on the propulsion plant, it is not impractical as will be shown in the following sections.

Transport Factor for 1500-ft, 50-kt Displacement Ship

Using the above-calculated R_t and a displacement of 63,800 LT, the transport factor is now calculated to be:

$$TF = \frac{\text{Ä}}{R_t} \times OPC = \frac{63{,}800 \times 2240}{2{,}310{,}000 \times 1.10} \times 0.65 = 37$$

This is the value plotted on Figure 3 and clearly demonstrates the significantly higher efficiency of a very long displacement ship compared to other marine forms when operating at 50 kts. The practical installation of large power sources in a limited size hull is discussed in subsequent sections of this paper which deal with the naval architecture aspects of this concept.

Seakeeping Characteristics, Head Seas

Criteria for judging the acceptability of a high-speed ship include both low resistance per ton of displacement and "good" seakeeping. The former has already been established for the present ship concept and the latter is now discussed.

"Good' seakeeping requires low heave and pitch motions since wave added drag, slamming, deck wetness, and bow immersion all increase with increasing motions. Analytical studies and experimental results show that the motions of a ship in a seaway are dependent upon the static ship-wave geometry (ratio of wave length to ship length) and upon the dynamic effects associated with the coupled ship-wave system. These dynamic effects can magnify the ship motions. Published literature contains numerous excellent studies of seakeeping, which are familiar to most naval architects and, hence, will not be reviewed in this paper.

A fundamental conclusion in these seakeeping studies is that, for wave lengths shorter than 0.75 LWL, there is very little ship response at any speed. As stated in [4], for these relatively short waves, "the exciting forces and moments are too small to cause appreciable motion so that the likelihood of deck wetness, high accelerations and slamming is also small". It will be shown that the 1480-ft LWL ship proposed in this study is particularly suited to take advantage of this criteria. Consider the 1480-ft ship running in head seas of significant wave height $H_{1/3}$ = 30 ft. This combination can be considered to be a modestly severe operating environment. Using the Pierson-Moskowitz sea spectrum relations the following major properties of this seaway are defined:

T_m	=	2.76 (H 1/3)$^{1/2}$ = 15.1 sec
ω_m	=	2 x 3.14/T_m = 0.42 rad/sec
L_m	=	5.12 x T_m^2 = 1170 ft
L_m/LWL	=	0.79.

The ranges of wave periods which contain measurable wave energy are:

$$5.5 \text{ sec} \le T \le 17.4 \text{ sec}$$

$$155 \text{ ft} \le Lw \le 1550 \text{ ft}$$

$$0.10 \le \frac{Lw}{LWL} \le 1.05$$

where:

T_m = modal period of wave spectrum, sec, (period of maximum energy)
ω_m = circular frequency of wave having the modal period, rad/sec
L_m = length of wave have the modal period Tm, ft
T = wave period, sec

L_w = wave length corresponding to period T, ft
LWL = load waterline of ship =1480 ft.

It is seen that at the period of maximum wave energy (15.1 sec) the corresponding wave length is only 0.79 LWL. Thus, there will be relatively little response of the ship to this wave at any speed, even at the resonant frequency of encounter. (In the present case it is estimated that the natural pitch and heave periods of the 1480-ft ship at 30-ft draft are approximately 7.3 secs. At 50 kts in a head sea, the period of encounter with the modal wave is 7.2 secs. Hence, although at resonance, and thus, amplification of ship motions, is expected, the wave forcing function is so small that only small motions are expected.) For all wave periods between 5.5 and 15.1 secs the length of the corresponding waves are substantially smaller than 0.75 LWL so that the wave induced hull motions will be small. At the largest wave period of 17.4 secs the wave length is 1.05 LWL. While this wave should be capable of developing ship motions, the wave energy in this component is zero so that no motions will occur.

The above qualitative explanation further demonstrates the advantage of long ships, not only is the hull efficiency high, but also the seakeeping characteristics are expected to be most satisfactory. Future studies of this concept will attempt to calculate the ship motions for a range of sea states and ship headings.

While model tests of the proposed hull have not been conducted, some measure of the expected seakeeping performance can be obtained from the model tests of the MARAD hull series [2]. From Table 2 of [2], a model hull with dimensions, loading and speed most closely equal to those of the present concept was selected as a “parent” for the seakeeping estimate, Table 1.

Table 1. Comparison of MARAD and Present Hull

	MARAD Hull	Present Hull
LWL, ft	1381	1480
Beam, ft	113	117
Draft, ft	30.3	30
Cb	0.59	0.44
$\Delta/(0.01\ \mathrm{LWL})^3$	30	20
LWL/Beam	12.2	12.6
Δ, LT	80,000	63,800
Vk (max.), kts	50	50

The major difference between hulls is that the present hull is more lightly loaded . According to the conclusions in [2] the lighter loading will reduce hull motions so that using the MARAD results will be conservative. The MARAD hull was model tested in the following equivalent head sea-state for speeds up to 50 kts.

H1/3 = 35.4 ft
Tm = 16.4 secs.

A summary of the test results for this hull follows.

Deck Wetness: There was no water shipped over the decks. This is a clear advantage of a low displacement-length ratio hull and arises because of reduced motions and the larger absolute freeboard of the long slender hull.

Average Bow Immersion: This is defined as that part of the immersion-emergence cycle in which the oncoming wave encounters the bow above the static waterline. It was found that the bow immersion was only 20% to 25% of the bow freeboard. This is consistent with the lack of deck wetness.

1/10th Highest Bow Acceleration: It was found that this was approximately 0.20 g. Although not measured, the center-of-gravity accelerations are expected to be less than 0.20 g.

Slamming Impacts: At 50 kts there was just one "moderate" slam in 37 cycles. Unfortunately, the term moderate was not defined.

The above discussion of head sea operation is clearly of a qualitative nature, but is based upon realistic physical relations which are to some degree substantiated by test results. In conclusion, it is not expected that operation in a 35-ft significant wave height head sea will be a "show stopper" so that further studies of long slender hulls are justified.

Transverse Stability

Intact Stability: The initial intact roll stability is measured by the metacentric height, (GM), in the upright position. Referring to Figure 5, the following equation can be written:

$$GM = KB + BM - KG$$

where:
GM = metacentric height
KB = height of center of buoyancy above baseline
= draft x constant
BM = metacentric radius
$= \dfrac{(\text{beam})^2}{\text{draft}} \times \text{constant}$
KG = height of center of gravity above baseline.

It is to be noted that GM is independent of ship length. For a given draft, KB is fixed and BM increases as the square of the beam so that the large beam (117 ft) in the present design is most advantageous in increasing the roll stability of the ship. From hydrostatic calculations for the 63,800 LT ship, the following values are obtained:

KB = 17 ft
BM = 35
KG = 38 ft

Thus: GM = 17 + 35 - 38 = 14.0 ft.

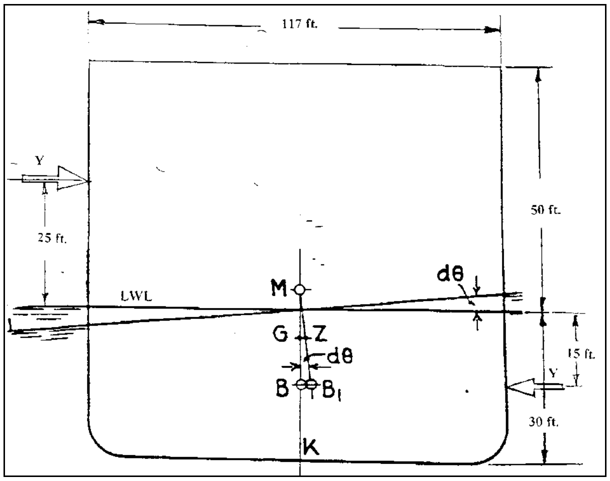

Figure 5. Location of Centers of Buoyancy and Metacenter and Side Force Due to Wind

This is 12% of the beam (117 ft) which is substantially larger than exist for most commercial ships and, hence, implies adequate static roll stability.

Roll Angle Due to Beam Wind: Using an 80-kt beam wind blowing against the side of the hull, which is assumed to 80-ft deep, (see Figure 5) and 1500 ft LOA, the side force is:

$$Y = (1/2)\,\rho\,(80 - 30) \times 1500 \times Cd \times (V_k \times 1.69)^2$$

Where:

ρ = mass density of air
Cd = aerodynamic drag coefficient normal to hull side = 1.00
V_k = wind velocity = 80 kts
Y = side force on hull due to beam wind.

Neglecting the sail area of the superstructure (expected to be small relative to the side area of the hull) the wind generated side force is:

$$Y = 1{,}714{,}000 \text{ lb} = 765 \text{ LT.}$$

The roll moment, Mr, due to this wind force being resisted by an equal and opposite hydrodynamic side force on the submerged portion of the hull, Figure 5, is:

$$Mr = Y\left(\frac{50}{2} + \frac{30}{2}\right) = 30{,}600 \text{ ft tons.}$$

The equilibrium roll angle is calculated using the previously estimated GM = 14 ft for this ship:

$$Mr = GM \times \sin\theta \times 63{,}800$$

So that:

$$\sin\theta = 0.034$$

and

$$\theta = 2.0 \text{ degs.}$$

This small roll angle assures a large reserved stability so that additional roll disturbances such as off-centerline loading; hull flooding due to damage; and hull motions during maneuvering can be tolerated without excessive listing or capsizing.

Rolling in Waves: The natural roll period, Tr, of the present design can be approximated by the following equation:

$$Tr = 0.44\,B / GM^{1/2} = 13.8 \text{ secs.}$$

For a beam sea having a 30-ft significant wave height, it has been shown that the modal period is 15.1 secs. Further, using the Pierson-Moskowitz sea spectra relationships, the significant wave slope is 4.6 degs. Assuming a roll damping factor = 0.20 critical for a hull with bilge keels and a linear dynamic system in roll, the magnification factor for $T_m/Tr = 1.09$ is approximately 5.0. Thus, the significant roll angle of the hull in this beam sea-state is 4.6 x 5 = 23 degs. When applied to the geometry of the hull having an 80-ft hull depth, the rolled down deck edge will be approximately 25 ft above the water surface.

This simple analysis indicates that rolling in beam seas is not expected to be a problem. Future studies should apply computer-based simulations and model tests for a range of sea states and ship speeds to further quantify dynamic roll behavior in a seaway.

Coursekeeping and Maneuverability

Coursekeeping implies maintaining a steady course with a minimal activation of controls and is primarily dependent upon the hull geometry and proportions. Maneuverability is related to changing directions of motion (turning or course change) and is dependent upon the magnitude of the control forces applied by rudders; transverse thrusters; deflected waterjets; etc. Analytical methods for evaluating these characteristics require a combination of theoretical analysis, experimental results, and operational experience. For conventional ship proportions methods do exist for making reliable estimate of coursekeeping and maneuvering. Indeed, computer-based maneuvering simulators are widely used throughout the world.

Unfortunately, the dimensions, proportions, and speed of the present 1500-ft hull are on the fringes of the limits of application of existing methods and so preclude their direct application to this concept. Nevertheless, the following qualitative conclusions can be offered based upon reasonable extrapolation of published results.

Figure 6, based on the results of [5], presents a plot of control fixed dynamic stability boundaries for a displacement ship as a function of length-beam ratio, block coefficient, and beam to draft ratio. For the present design, LWL/B = 12.7, C_b = 0.43, and B/H = 4.0. The combination of these parameters is beyond the boundaries of Figure 6, but is in a direction of assuring dynamic stability for the present high length-beam ratio concept. This conclusion is confirmed by the observations in [6], where it is stated that increasing LWL/B improves dynamic stability and increases the time to change heading.

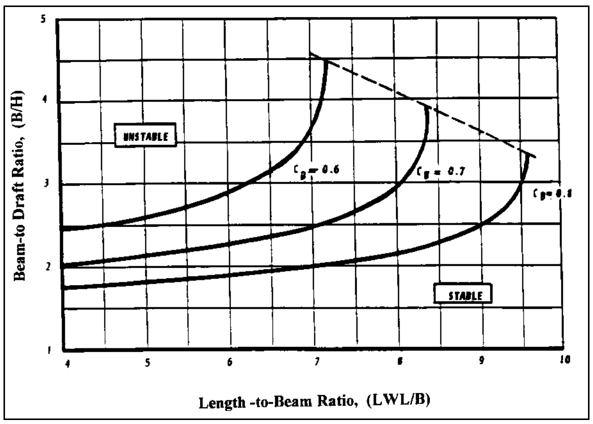

Figure 6. Dynamic Coursekeeping Stability Boundaries

Relative to maneuvering, for a given steering moment about the center of gravity, the turning radius increases with increasing LWL/B [5 and 6]. Since the present design anticipates the use of deflected waterjet thrusters and bow thrusters to provide the steering forces, it may be that the turning radius can be controlled to attain reasonable values. Although high-speed turns are anticipated, the resultant roll angle is expected to be small because of the large GM associated with the relatively large beam of this design.

Maneuvering this long ship in harbors is, of course, of concern and should be evaluated in future studies. This, and operation in deep water, will require extension of present analytical techniques and collection of additional model test data using rotating arm or planar-motion test facilities. These data should be obtained for a wide range of hull parameters representative of long slender ships; speed; and water depth.

In summary, it appears that this long ship will be dynamically stable, but will require relatively larger steering forces (compared to convention displacement hulls) to provide reasonable maneuvering characteristics.

CONCEPT DEVELOPMENT

Design Synthesis Model

Having established the "hydrodynamic viability" of the fast displacement hull concept using the MARAD and Series 64 model tests, a conceptual design was needed so that the "engineering viability" could be assessed. The primary focus of this concept design was to determine if such a hull could support the propulsion plant, fuel and payload, while still meeting basic structural and stability requirements. This concept design was developed using a design synthesis model called PASS.

Since the combination of size and speed is currently far outside the norm of current ship designs, synthesis models that are based on historical data are not particularly suitable for this application. The PASS (Parametric Assessment of Ship Systems) model is unique inasmuch as it uses, to whatever extent practical, algorithms derived from first-principle physics rather than from empirical data to characterize all major subsystems and their relationship to the overall ship. This ensures that new technologies are realistically modeled without being biased by existing (and maybe out-dated) trends in ship or ship-subsystem design. In no case, however, are the algorithms completely without empiricism. The design synthesis "engines" integrated into PASS have been used and tested extensively over the years on numerous ship programs.

The design process employed utilizes the traditional design spiral. The design is initiated with the user input that describes the basic hullform to be used. This is accomplished, for monohulls, by defining the midship section coefficient, a range of waterline lengths to be evaluated, a range of block coefficients to be evaluated, and a range of length-beam ratios to be evaluated. A conceptual hullform and a set of hydrostatics are first developed for each combination of length, length-beam ratio, and block coefficient.

Next, global and local loads are developed and combined with material properties to determine the required scantlings size. Details of hull decks are also developed. This description of the hull structure is then used to develop an estimate of the hull weight.

Once the structure is defined, a drag estimate is made and the propulsion plant and propulsors are sized. The user must define the type of propulsion plant, i.e., diesel, gas turbine, electric, CODOG, etc., and the type of propulsors, i.e., fixed-pitch propeller, variable pitch propeller, waterjet, etc. A check is then made to determine if the estimated engine room size will fit within the hull. A weight estimate is also developed for this propulsion plant.

The electric plant design is developed in a manner similar to the propulsion plant design. Again, the user can specify the type of generator prime movers or the specific number and sizes of generators. An electric load analysis is developed.

Algorithms for auxiliary systems and outfitting determine the weight and volume requirements for these systems and are based on crew size, ship size, levels of automation, etc.

Once the lightship configuration is developed, fuel loads are calculated, payloads totaled, and a full-load displacement is calculated.

Design

Since the hullform characteristics for the high-speed ship are unique, as described earlier, no variation in the hullform was made during this study. The length, length-beam ratio, and block coefficient previously calculated were input directly to the program. Additionally, other details of the Series 64 hulls were input to improve the definition of the hull. The only geometry variable was the draft, which is an output of the calculations. Since the draft is also crucial to the performance, both the payload and range (i.e., fuel load) inputs were varied to control draft. The characteristics of the design generated by PASS are provided in Table 2 and closely match those of hulls previously described.

Hydrostatics and Intact Stability: As already discussed, the hullform is fixed by the characteristics of the parent model hullforms. As part of the design process an intact stability analysis was made by calculating a righting arm curve and assessing the stability compared to U.S. Navy standards. Table 3 shows the results of the intact stability analysis for various loading conditions. Figure 7 provides the righting arm curve.

As shown in Figure 7 and Table 3, the design has more than sufficient intact stability through the life of the ship. Design and growth margins are included in the KG estimate. A damaged stability analysis has not yet been done because it is outside the scope of this current study. It is anticipated that significant subdivision may be necessary. Additionally, the length of this ship could cause extreme trim angles when compartments at the bow or stern are damaged.

General Arrangements: Weight distribution will be critical to this design to maintain the proper running trim and draft. As will be seen in the following sections, this design requires a large, heavy propulsion plant and a large fuel capacity.

It will be necessary to locate the variable loads (fuel), as close to the center of gravity as possible. In addition, some fuel will have to be carried fore and aft of the center of gravity to provide some trim control.

Table 2. Concept Design Characteristics From PASS

Length Overall, ft	1517.7
Length, Waterline, ft	1480.0
Beam, Maximum, ft	117.1
Beam, Waterline, ft	117.1
Draft, ft	29.03
Depth, ft	79.03
Displacement, Full-Load, LT	63,796
L/B	12.64
C_b	0.440
Design Speed, kts	50
Range at Design Speed, nm	9000
Endurance, days	20
Main Engines	10 GTs at 57,000 hp
Propulsors	5 Inducer Waterjets
Electric Plant	Four 2400 kW SSDG
Weights, LT	
Structure	23,957
Propulsion Machinery	5,237
Electric Plant	774
Electronics	25
Auxiliaries	2,812
Outfitting	1,295
Fuel	16,100
Cargo	12,000

Table 3. Stability Results From PASS

	Full Load	End of Service Life	Min Op	Min Op With Ice
Displacement	63,791	66,985	41,067	42,923
Area Ratio[1]	6.84	6.99	3.26	3.32
GM (Transverse)	14.34	11.67	14.94	11.36
GZ Ratio[2]	0.11	0.13	0.33	0.14

1. Minimum Acceptable 1.4.
2. Maximum Acceptable 0.6.

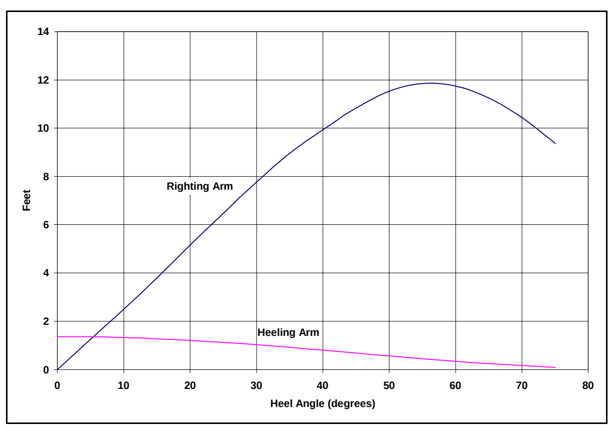

Figure 7. Righting Arm Curve Output of PASS

The cargo holds will also have to be distributed fore and aft so as not to result in extreme trim angles when the ship is in its ballast condition in port.

Figure 8 shows a profile of what the ship arrangement might look like.

Initially, it has been assumed that this would be a container ship, however, there is sufficient volume so that it could also easily be a RO/RO or bulk cargo ship.

Propulsion Plant: Based on the speed requirement and the power needed, two propulsion plant variations were examined using PASS. The first variation utilized single gas turbine engines each matched to single, mixed-flow, KaMeWa type waterjet. The second variation explored the use of inducer type waterjets each driven by a pair of gas turbine engines.

For the first variation, the propulsion plant consists of nine gas turbines each driving a mixed-flow waterjet. The waterjets are rated at approximately 67,000 hp each. Each gas turbine/waterjet combination is similar to what is currently under consideration for the Fast-Ship Atlantic project. The FastShip design plans on utilizing 67,000 hp Rolls Royce gas turbines driving KaMeWa waterjets. The propulsion plant will have to be split between two engine rooms. The forward engine room will house five gas turbines and the aft engine room four. This split arrangement is necessary to provide adequate room for maintenance and propulsion auxiliaries. 67,000 hp waterjets will have an installation diameter at the transom of approximately 16 ft. Nine such waterjets will require a stacked style installation to fit in the transom width. This will result in decreased efficiency. Another disadvantage is that the upper level waterjets may require an auxiliary system to prime them. This arrangement will also require some development effort, however, these issues have already been examined on smaller scale installations.

This nine across engine arrangement will also result in a tight fit for the outboard gas turbine engines. These engines are likely to be close to the hull sides and may require a unique offset gearbox configuration to move them inboard to provide for working room around them.

Steering would be accomplished by the outer two waterjets on each side, the inner five jets would not have steering buckets, however, they may require reversing buckets to meet regulatory issues for crash stops.

During the operation of this ship, only the required number of propulsion plants would be brought on-line as needed. During in-shore and harbor operations, where speed would be reduced, some plants would be shut down to conserve fuel.

The second propulsion plant variation consists of five inducer waterjets each driven by a pair of 60,300 hp gas turbines, 10 gas turbines total. This propulsion configuration is currently being designed by Band, Lavis & Associates, Inc. for another project. Each of the waterjets would be rated at 120,000 hp. Each pair of gas turbines would be joined by a combining gearbox to a single waterjet. As with the first variation, only the outer 2 waterjets would have steering buckets.

The advantages of the inducer waterjet installation is that the installation diameters are significantly less, approximately 13 ft.

Although an inducer waterjet is generally lighter than a mixed-flow jet of the same power rating, the fact that these jets are twice the power rating of the mixed-flow waterjet configuration offsets that weight advantage somewhat. There will be some weight savings compared to the fixed-flow waterjet configuration, however, at this stage it has been assumed that this weight savings will be consumed by the additional gas turbine and more complex reduction gears.

The major disadvantage to the inducer configuration is that each waterjet will be driven by two gas turbines through a combining gearbox. As with the mixed-flow waterjet plant, the outboard gas turbines will be very close to the hull. This problem can be overcome with a stacked engine configuration using a vertically oriented reduction gear rather than a traditional side-by-side configuration. For this installation the waterjets will not have to be stacked. The major disadvantage for the inducer configuration is that the waterjets will require a significant design and development effort to bring them into production. The gas turbine engines will not require any such development.

Due to the number of waterjets, their size and arrangement, the inlet design and configuration for both propulsion plant options will have to be examined closely in future studies. The large size and arrangement could result in cavitation problems due to their close proximity to each other and the interface with the hull shape. Bifurcated inlets will certainly have to be considered. The optimum hull shape for the installation may not be fully compatible with the basic hull configuration. A detailed CFD analysis and hullform optimization study will be needed.

In conventional ships it would be expected that at 50 kts, cavitation could be a potential problem associated with support struts, rudders, propellers, shafts, brackets, etc. Fortunately, in the present design the ship is propelled by waterjets which are deflected to provide steering forces so that none of the usual cavitation prone appendages exist.

The stacked configuration for the mixed-flow waterjets will need to be studied carefully to determine what the additional loses and inefficiencies are. Due to this potential decrease in efficiency and resulting increase in fuel load, the inducer waterjet propulsion plant has been used for this notional design.

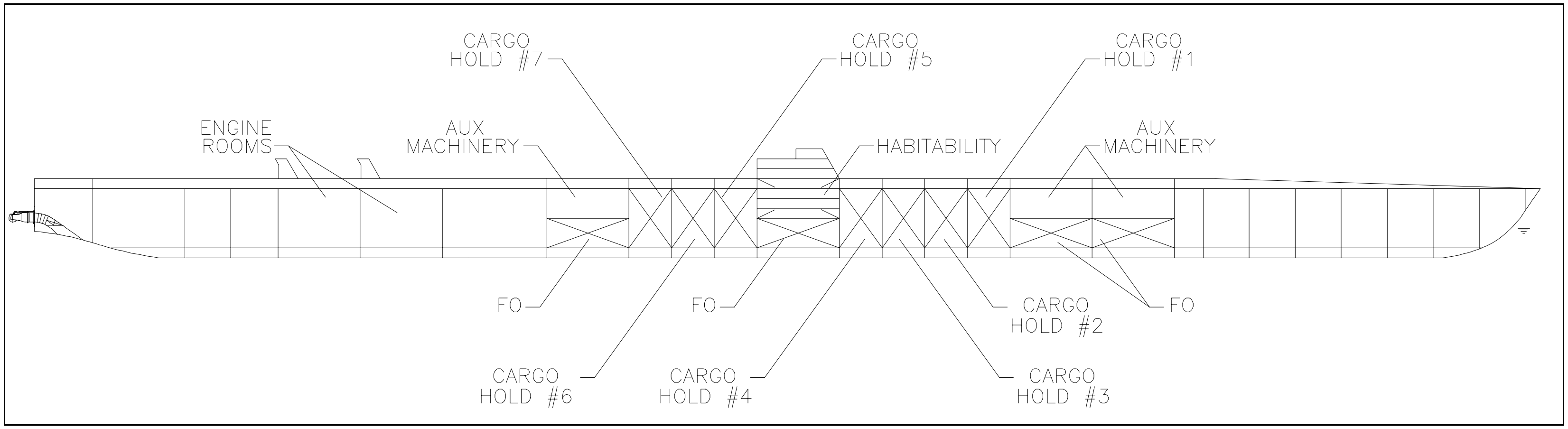

Figure 8. Inboard Profile

Electric Plant: The electric plant is a basic diesel generator system and has been assumed to consist of three 2400 kW generators and one 2400 kW standby generator. There is nothing special or unique about this system. Power has been provided assuming that there may be a large number of refrigerated containers. It is anticipated that a high voltage system with localized transformers would be employed to minimize weight and cost. The SSDGs will most likely have to be distributed forward in the ship (along with cargo and tankage) to help compensate for the aft weight of the propulsion plant.

Outfitting and Auxiliaries: Outfitting and auxiliaries are based on a crew size of 25 and an endurance of 20 days. Standard berthing, messing and lounge facilities would be provided. However, because of the excessive volume of the ship and the significant impact aerodynamic drag will have, the deckhouse size will be minimized. This will put the crew in the hull. This will also place the crew closer to the center of gravity providing a more comfortable ride. However, few, if any, staterooms would have windows.

Structure: The sizing of the midship section is based on a bending moment calculation using the traditional trochoidal wave with a wave height of 1.1 x $(LWL)^{1/2}$ and wave length equal to LWL. This calculation has not been altered for this design. It has been assumed that this size wave, which may be rare in the real ocean environment, will in fact, be present at all times while it is operating at speed-length ratios of approximately 1.3. The initial estimate shows a primary structure configured as follows:

Inner Bottom Height	10.5 ft
Average Deck Height	9.0 ft
Bottom Plate	2.56 inches
Average Bilge Plate	2.19 inches
Average Side Plate	1.18 inches
Average Sheer Plate	2.00 inches.

The single most significant characteristic that effects the structural arrangement for this design is the hull depth. In general, as a ship grows in size, at some point the freeboard does not have to increase proportionally with the length to maintain a dry ship with good seakeeping qualities. This is the case with the current concept. A 50-ft freeboard is more than adequate for the type of ocean service this vessel will see. However, this freeboard is more appropriate for a 1000-ft ship. Thus, the depth-to-length ratio of this large ship is quite low. This results in a very low section modulus.

Based on the basic structural concept developed by PASS, ABS used a specially modified version of SafeHull for containerships to design midship sections for the appropriate principal data and loads. This initial scantling evaluation confirmed the plate thickness and structural weight estimates made by PASS and that the section modulus was critical to the structural design. Additional structural configurations were then run as described in the following section.

SAFEHULL EVALUATION

A preliminary structural design was performed using a specially modified version of the American Bureau of Shipping's SafeHull system. SafeHull is a first principles-based, design-oriented system for the assessment of the hull structure for certain vessel types. The SafeHull system is normally applied to vessels of conventional size and hullform, and has been developed for tankers, bulk carriers, and containerships.

SafeHull analysis is conducted in two phases. In Phase A the initial scantlings are checked and minimums for these scantlings are established. Phase B analysis is more detailed and requires a finite-element analysis to verify the strength, the maximum stresses (load effects) of the structure. In the present case, only Phase A as been applied.

For conventional vessels of the kind mentioned above, there is a history of successful performance that is implicitly captured in the SafeHull system. In the present case, several aspects of the design are outside historical experience. This applies particularly to the slenderness and speed of the vessel. Hence, the structural design presented below, is a result of the application of a specially modified version of SafeHull and should be regarded as tentative.

Two midship designs were considered. The first is an open deck design that is a variant of conventional designs, and the second is an unconventional closed-deck containership design.

Open-Deck Design

A conventional containership design has an open deck with deck box girders, as seen in Figure 9.

In the present design a partial deck structure is provided in the form of a three-cell box girder structure on either side. The deck box girder structure was located above the main deck in order to satisfy section modulus requirements. Although not investigated in the present design, a more conventional, entirely open design may provide sufficient modulus away from the midship region.

The wing and double bottom tanks may be used as water ballast tanks or fuel tanks. The cargo hold is located as it would be in a conventional design. As such, where the cargo hold is partially covered, it may be necessary to develop a special container-handling system.

The primary scantlings of the open deck design are listed in Table 4.

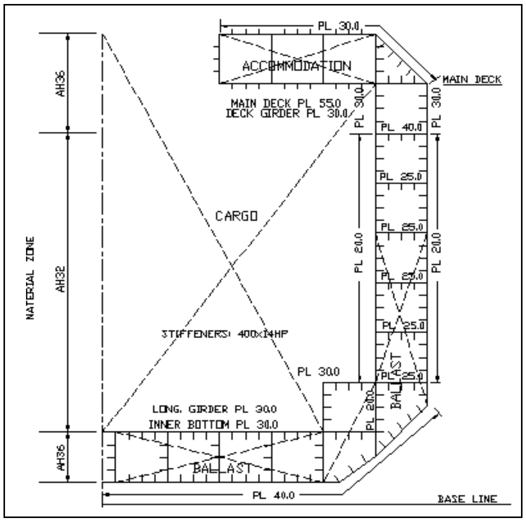

Figure 9. Open Deck Midship Section Design

Table 4. Primary Scantling for Open Deck Design

	Inches
Bottom Plating	1.58
Bilge Plating	1.58
Side Shell Plating	0.79
Sheer Plating	1.18
Main Deck Plating	2.17
Inner Bottom Plating	1.18
Inner Skin Plating	0.79
Bottom Girder Plating	1.18
Side Stringer Plating	0.98
Deck Box Girder Plating	1.18
Second Deck Plating	1.58
Deck Box Girder Plating	1.18

A total steel weight is estimated as 22,840 LT based on the midship section scantling.

Closed-Deck Design

From the structural performance viewpoint, there are certain advantages of closed-section designs over those of open-section. This applies particularly to hull girder response to torsional loads. Modern containerships are configured as they are for reasons of ease of loading and unloading.

As an alternative, a closed-deck design with intermediate decks was considered, see Figure 10.

The closed deck design has a larger section modulus than the open deck design. A multi-cell deck box girder was located below the main deck.

Clearly, an entirely new loading and unloading system is required as this design has some of the characteristics of a RO/RO vessel. As with the open-deck design it may be possible to provide openings away from the midship region. In this case a mixed system may be required whereby containers are loaded vertically in open areas and then moved to central areas of the vessel horizontally. These issues are outside the scope of this paper.

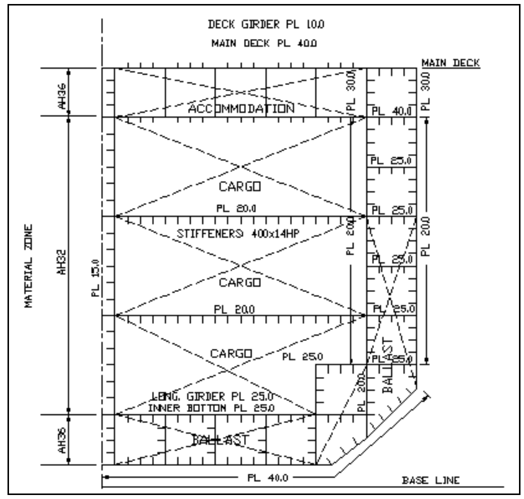

Figure 10. Closed Deck Midship Section Design

The primary scantlings of closed deck design are listed in Table 5.

Table 5. Primary Scantling for Closed Deck Design

	Inches
Bottom Plating	1.58
Bilge Plating	1.58
Side Shell Plating	0.79
Sheer Plating	1.18
Main Deck Plating	1.58
Inner Bottom Plating	0.98
Inner Skin Plating	0.79
Bottom Girder Plating	0.98
Side Stringer Plating	0.98
Second Deck Plating	1.58
Deck Box Girder Plating	0.39
Deck Plating	0.79
CL Long. Bulkhead Plating	0.59

A total steel weight is estimated as 26,710 LT based on the midship section scantling.

The steel weight for the closed-deck configuration is approximately 17% higher than the open deck configuration. However, it is also expected to be stronger as well. Optimization of the scantlings of the closed-deck configuration in future design efforts may reduce this weight.

Thus, in the future alternate configurations will have to be examined to increase the section modulus while minimizing the impact on ship weight and loss of range or payload. There are a number of ways the section modulus may be increased: (1) increase the hull depth locally near midships, (2) enclose the upper deck and find an alternate means of loading containers (this would not be a problem for a RO/RO configuration), (3) use larger than normal box girders at the sheer and also on the centerline, and (4) utilize a combination of all three options.

CONCLUSIONS

Previous model test programs have demonstrated the "hydrodynamic" viability of large, 50-kt displacement hulls. Interest in such hulls has been shown recently as stated in [1] and also with interest in the Fast-Ship Atlantic project. The obvious next step is to prove the engineering viability of such a concept. This study has taken that next step to determine if basic naval architectural, hydrostatic, structural and propulsive issues could be satisfied.

It has been demonstrated that the basic concept is viable, but not without some technology development and special design considerations.

A "balanced" design was found for the 1500-ft ship configuration, however, in the process of evaluating this design a number of R&D issues were identified. At this stage it does not appear that any of these issues are fatal to the concept, but further evaluation and design work is required.

Currently, this study is proceeding with the further development of the design to incorporate revisions to the structural configuration and the machinery arrangement.

It has not been the intent of this paper to present a viable economic solution. There are many operational issues that need to be addressed before that can be done. These issues include, but are certainly not limited to, harbor and port facilities for such a large ship, loading and offloading issues, regulatory issues for large, high-speed ships, drydocking and maintenance.

Some of the redesign and R&D issues that must be examined in more detail in the next design cycle are:

- Hydrodynamic
  - Seakeeping
  - Maneuvering
  - Damaged Stability
  - Waterjet Inlet/Hull Interface
- Structure
  - Dynamic Loads
  - Midship Section Design
- Propulsion
  - Large Gas Turbine Development
  - Large Waterjet Development
  - Reduction Gear Design
  - Waterjet Installation Configuration.

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